

Gas Stirling engine μ CHP boiler experimental data driven model for building energy simulation



J.-B. Bouvenot^{a,*}, B. Andlauer^b, P. Stabat^b, D. Marchio^b, B. Flament^a,
B. Latour^a, M. Siroux^a

^a ICube UMR 7357, Université de Strasbourg, INSA de Strasbourg, 24 Boulevard de la Victoire, 67 084 Strasbourg Cedex, France

^b Mines Paristech, Centre d'efficacité énergétique des systèmes, 60 Boulevard Saint-Michel, F 75 272 Paris Cedex 06, France

ARTICLE INFO

Article history:

Received 21 May 2014

Received in revised form 21 July 2014

Accepted 14 August 2014

Available online 24 August 2014

Keywords:

μ CHP

Micro cogeneration

Stirling

Modelling

Experimental

ABSTRACT

A dynamic model for the simulation of gas micro combined heat and power devices (μ CHP boilers) has been developed in order to assess their energy performances. From a literature review and experimental investigations, the new model is designed with the aim of limiting the number of parameters which need to be easily accessible in order to be suitable with annual building energy simulations. At first, this paper focuses on the description of the μ CHP boiler which has been tested, on the development of the test bench and on the experimental results. Then, it focuses on the model description, on its parameterization and on its validation. The modelling approach is based on an energy balance on the device and on empirical expressions for the main inputs and outputs. The model characterizes the μ CHP boiler behaviour for different inlet water flow rates and temperatures. The dynamic phases with the start-up and cooling phases are taken into account. Finally, the models for the Stirling engine and the additional boiler are limited respectively to 28 and 24. Further experimental investigations led to simplify the μ CHP model from 28 to 17 parameters without reducing the accuracy.

© 2014 Elsevier B.V. All rights reserved.

1. Introduction

μ CHP devices are high efficiency solutions for heat and electricity production which can reduce their environmental footprint by decreasing CO₂ emissions and by converting more primary energy than separate electricity and heating productions [1–4]. Moreover, μ CHP devices can be a solution for demand-side management to relieve the power grid and to avoid using high carbon power generation during the electrical demand peaks as shown by Vuillecard et al. [5]. CHP is already widespread in the industry and the district heating networks with a power range of MW. μ CHP corresponds to low power CHP (less than 15 kW_{el}) in order to be suitable for residential or commercial heating and power needs. Because of the low power output of μ CHP devices, the technologies which are used are different from centralized power plants [6]. The electrical efficiencies are also lower but the global efficiencies are slightly higher. The main conversion processes which are used for μ CHP applications are: micro turbines, Stirling engines, Ericsson engines, steam

engines (Rankine cycle), internal combustion engines and fuel cells [7]. Among these technologies, the study focuses on the gas Stirling μ CHP boilers which are increasing on the market (Table 1). μ CHP boilers consist in a Stirling engine sized to cover the heat and power base loads of an accommodation and an additional gas condensing boiler to cover the peak heating demands (space heating and/or domestic hot water (DHW)). The excess electricity production is exported on the public grid. However, their introduction is slowed down due to their high investment costs in countries where electricity prices are low. The development of models suitable for use in building energy simulations is required for the assessment of these μ CHP systems, notably in life cycle costs. Indeed, the potential growth of this technology on the market is highly dependent on investment costs, feed-in tariffs and investment subsidies.

The μ CHP boilers are generally controlled by the heating demand in order to maintain a high overall efficiency and to ensure the thermal comfort as a priority. The optimization of μ CHP solutions including the sizing of the auxiliary components such as the thermal storage requires the development of models able to represent correctly their performance, especially during the start-up/cooling phases.

* Corresponding author. Tel.: +33 03 88 14 49 67.

E-mail address: jean-baptiste.bouvenot@insa-strasbourg.fr (J.-B. Bouvenot).

Nomenclature

c	heat capacity, $\text{J K}^{-1} \text{kg}^{-1}$
e_{air}	air excess, –
GHV	gross heating value, J kg^{-1}
H	enthalpy, J mol^{-1}
$\dot{H}$	enthalpy flux, W
LHV	low heating value, J kg^{-1}
L_v	latent heat, J kg^{-1}
M	molar mass, kg mol^{-1}
$\dot{m}$	massflow rate, kg s^{-1}
$[MC]$	thermal mass, J K^{-1}
p	pressure, Pa
P	power, W
$\dot{Q}$	heat flux, W
t	time, s
T	temperature, $^{\circ}\text{C}$
$[UA]$	exchange coefficient, W K^{-1}
V_{air}	combustion air volume, $\text{N m}^3 \text{N m}^{-3}$
V_m	molar volume, $\text{N m}^3 \text{mol}^{-1}$
$\dot{V}$	volume flow, $\text{N m}^3 \text{s}^{-1}$

Greek symbols

ΔT	temperature difference, K
Δt	time step, s
χ	molar fraction, –
η	efficiency, %
ρ	density, kg N m^{-3}
τ	time constant, s

Subscripts and exponents

0	reference
air	combustion air
amb	ambient
aux	auxiliary
cond	condensation
cool	cooling phase
cw	cooling water
el	electrical
electro	electronic
end	end
exh	exhaust gas
fuel	fuel
gross	gross part
HX	heat exchanger
i	inlet
int	internal
lat	latent
loss	skin heat losses
max	maximum
net	net part
nom	nominal
o	outlet
P	power
PV	process value
Q	heat
sen	sensible
SP	set point
start	start-up phase
stop	stop phase
th	thermal
tot	total

Besides, the model of μCHP boiler to be developed for building energy simulation tools should meet the following requirements:

- Short computation time in accordance with annual building energy simulations
- Accuracy in steady states and transient (start-up and cooling phases) conditions
- Easiness to parameterize

2. Modelling approaches

Many Stirling engine models have been developed for the description of the engine internal phenomena, and not for CHP applications coupled with the building. The usual time steps for accurate thermodynamic models (from the order of 10^{-3} to 10^{-6} s) are very different in comparison to building simulations (from 15 min to 1 h). Moreover, to correctly model an engine, many features have to be known like heat exchanger surfaces, work pressures, geometry, etc., which are generally unknown. Many thermodynamic models on Stirling engines exist: Schwendig et al. [8] modelled the global Stirling cycle, Kongtragool and Wongwises [9] introduced the dead volumes effects. Organ [10] first programmed CFD models of Stirling engines [11]. Urieli and Berchowitz [12] worked on a simplified thermodynamic model in order to find a compromise between accuracy and computational time. However, this last model still requires many data, mainly on the regenerator and piston geometry. As a consequence, these detailed models do not match to the problem of coupling with the buildings.

Beausoleil-Morrison and Kelly [13] have undertaken a literature review on models of mini- and μCHP tools for dynamic simulation of buildings. Among the models identified in the literature review, Beausoleil-Morrison and Kelly mention McRorie and Underwood [14] whose model would suppose substantial effort to be integrated in building simulation codes, Kelly [15] who models a very specific two cylinders diesel engine, and the model of Pearce et al. [16] which does not take into account seasonal variations of efficiency. The authors [13] conclude that none of these approaches is suitable for coupled modelling to the building, since these models cannot be used to study control strategies and thermal storage which play an important role in the integration of μCHP technologies in buildings [17].

Furthermore, “black box” models have been developed. Thiers et al. [18] have fitted a quasi-static model of a wood pellet Stirling engine from a measurement campaign conducted on the Sun-machine. The start-up and cooling phases are well characterized. This model is strictly limited to the tested product and to the achieved test conditions (source temperatures, water flow rates, fuel characteristics, heating load). Another black box generic model is proposed in the French thermal regulation which uses a constant annual efficiency [19] assessed by a short time experimental test. The efficiency is independent on any operating conditions. Conroy et al. [20] made a numerical model of the gas Stirling engine μCHP Whispergen based on field tests. The accuracy of the measurements is not given and the fitting only depends on time without physical meaning. Roselli et al. [21] also studied the energy performances of 8 μCHP devices including 3 power modulating devices. Steady states have been characterized but no numerical model has been implemented.

The model of Annex 42 of the International Energy Agency [13] is the closest to our set objectives. It is adapted for the implementation of μCHP models (internal combustion engine and Stirling engine) in dynamic simulation piece of software. This model uses 2 thermal capacitance nodes. An optimization procedure is used to identify the 91 parameters but the uniqueness of the solution (identified parameters) is not guaranteed. Lombardi et al. [22] have

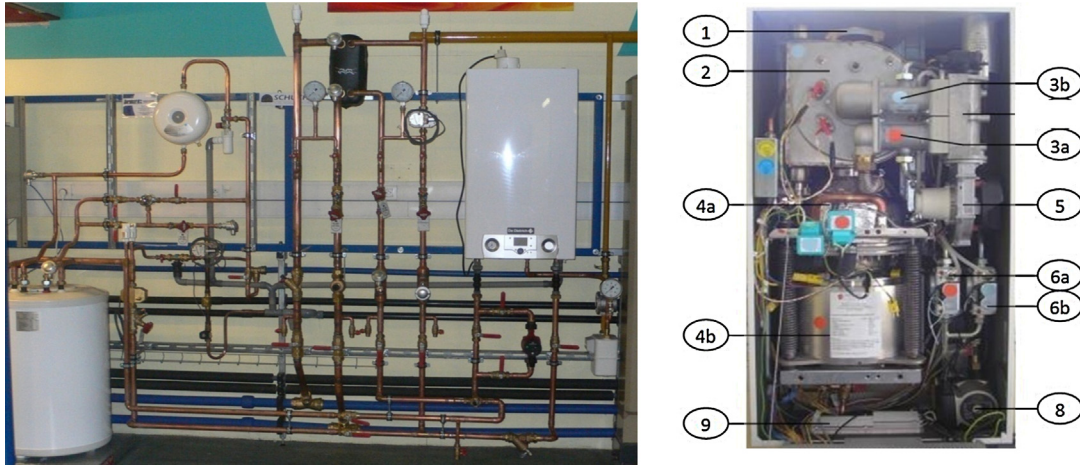


Fig. 1. Test bench (left) and μ CHP (right). * 1: exhaust gas outlet, 2: condensation heat exchanger, 3a & b: burners, 4a: head of the Stirling engine, 4b: Stirling engine, 5: fan, 6a & b: gas suppliers, 8: water pump, 9: DHW heat exchanger).

simplified this model by implementing a linear difference model based on linear coefficients and reference parameters which are determined by a few experimental tests. Borg and Kelly [23] also developed a dynamic model for tri generation studies, based on inflows and outflows of the system. They also underline the relevance to take into account the dynamic behaviour of energetic systems to obtain realistic results in comparison with static models.

This paper presents a generic model of μ CHP boilers, based on experimental investigations on a gas Stirling engine μ CHP system, with a few parameters easily reachable by simple experiments or published characteristics. Each parameter is determined independently without global minimization.

3. Experimental investigations

3.1. Experimental set up

The tested unit is a gas Stirling engine μ CHP from De Dietrich/Remeha called Hybris Power including an additional condensing gas boiler. Hereafter, both the Stirling engine and the additional boilers are separately studied. The test bench is designed

to measure the main values involved in the energy balance. The bench gets several operation modes:

- Instantaneous heating (heating and DHW) and electric production.
- Semi stored heating: a part of the heat production is stored in a storage tank of 130 l.

Figs. 1 and 2 present the bench and the metrological means:

The metrological means (Table 2) have been selected to provide acceptable uncertainties for the fuel consumption, electrical and thermal outputs, which are comparable with other experimental trials [24–26].

The thermal power is calculated by the thermal energy transport equation (Eq. (1)):

$$\dot{Q}_{HX} = \rho(T_{CW,i}) \cdot \dot{V}_{CW} \cdot (c_{CW}(T_{CW,o}) \cdot T_{CW,o} - c_{CW}(T_{CW,i}) \cdot T_{CW,i}) \quad (1)$$

The thermal properties of the water are obtained from standard tables. The fuel power is calculated by measuring the gas volume flow converted to normal conditions of pressure and temperature and by using the GHV of the gas daily given by the supplier.

$$P_{fuel} = \dot{m}_{fuel} GHV \quad (2)$$

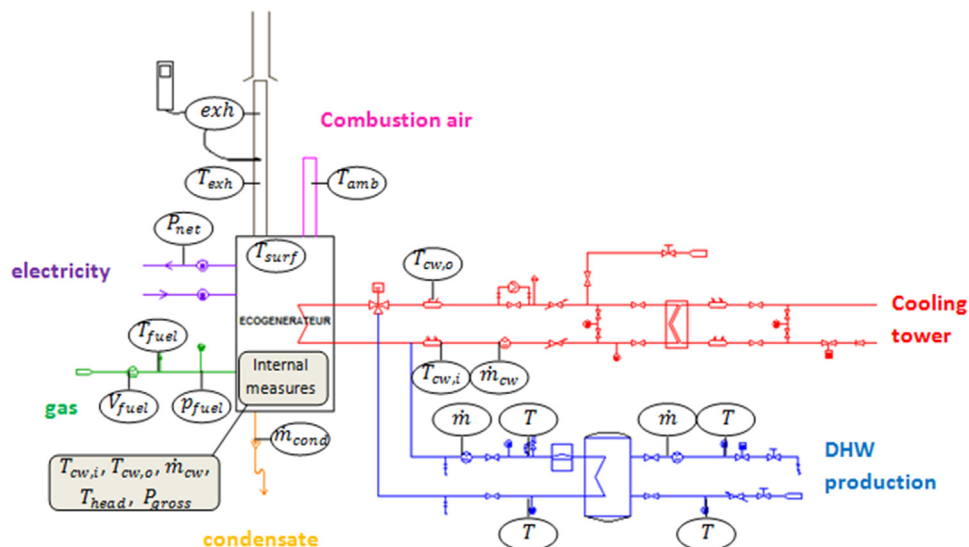


Fig. 2. Hydraulic principle scheme.

Table 1
Existing gas μ CHP prototypes.

	Manufacturer	Apparatus	Power		LHV efficiency	
			Electric, kW _{el}	Thermal, kW _{th}	Electric, %	Thermal, %
Gas Stirling engine (gas SE)						
	Enatec	Infinia	1	6.4	12.5	80
	SOLO	Stirling 161	2–9.5	8–26	22–24.5	65–75
	Disenco	Inspirit	3	15	16	76
	Baxi	Ecogen	0.3–1	3.7–25.3	16	83
	Viessmann	Witowin 300-W	1	3.6–20	15	82
	WhisperGen	EU1	1	7.5–14.5	11	84
	De Dietrich Remeha	Hybris Power	1	3–23.7	17	85
	Senertech	Stirling SE	1	3–23.8	14	77
	Sunmachine	Gas	1.5–3	8–15	25	65

● : available.

● : introducing phase.

The electrical power is measured by an energy metre which measures the incoming electricity consumed by the auxiliaries (fan, pumps, electronic card) and the out coming electricity produced by the Stirling engine. Uncertainties are calculated according to the propagation law of uncertainties [27,28].

3.2. Experimental results

3.2.1. Stirling engine in steady state

The main results have been plotted in Figs. 3–7. The tests show that the inlet water temperature has a significant influence on the electrical production, the condensation gains, the fuel power and

the exhaust gas temperature. The variations of the thermal power are low and are located in the range of the uncertainties (Fig. 3).

The theoretical Stirling cycle efficiency depends on the temperatures of the cold and hot sources. The inlet water temperature represents the cold source and the burner supplies the hot source of the engine. Low inlet water temperatures increase the temperature gap between the two sources and increase the theoretical efficiency. For the condensation gains, the heat exchanger surface has to be the coldest as possible to condensate the steam from exhaust gas. Lowest inlet water temperatures in the heat exchanger can achieve significant heat gains. The mass flow rate also has an

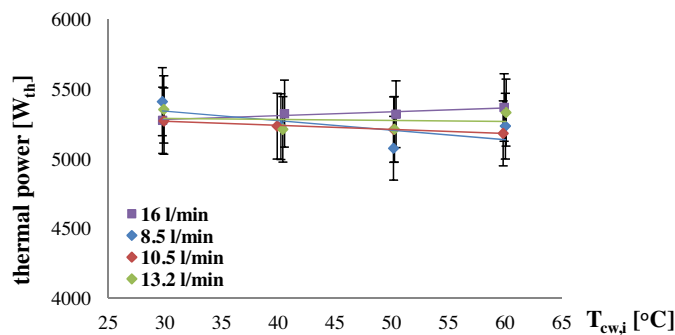


Fig. 3. Thermal power of the Stirling engine.

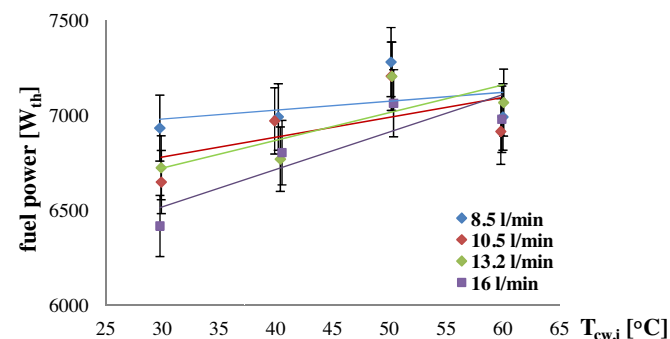


Fig. 4. Fuel power of the Stirling engine.

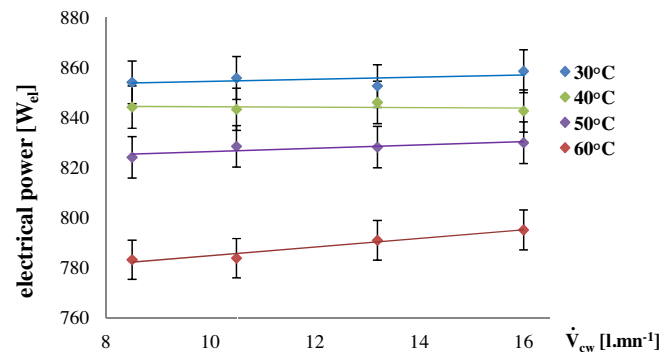


Fig. 5. Gross electrical power of the Stirling engine.

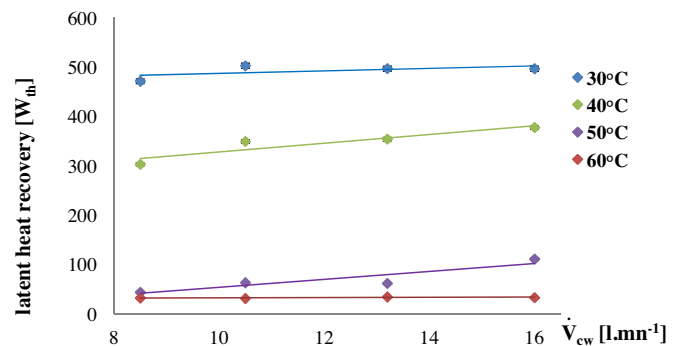


Fig. 6. Condensation gains of the Stirling engine.

Table 2
Metrological means.

	Type	Notation	Metrologic means	Error
Fuel	Consumed fuel	$\dot{V}_{fuel}$	Pulse counter	$\pm 1.5\%$
	Fuel pressure	p_{fuel}	Hand pressure metre	± 1 mbar
	GHV	GHV	Gas supplier data	± 0.1 kWh N m ⁻³
	Temperature	T_{fuel}	Pt100 probe	± 2 K
	Fuel power	P_{fuel}	Calculated	$\pm 2.5\%$
Heating output	Inlet water temperature	$T_{cw,i}$	Pt500 probe	± 0.2 K + 0.05%
	Outlet water temperature	$T_{cw,o}$	Pt500 probe	± 0.2 K + 0.05%
	Volume flow	$\dot{m}_{cw}$	Pulse counter	± 0.15 L mn ⁻¹ + 2.5%
	Thermal energy	$\dot{Q}_{HX}$	Calculated	$\pm 4.5\%$
Electric output	Electric power	P_{net}	Watt metre	$\pm 1\%$
	Temperature	T_{exh}	Pt100 probe	± 0.3 °C
Exhaust gas	Gas analysis: O ₂ , NO _x , CO, CO ₂ , THC or CH ₄		Combustion analyser	

influence because it plays on the mean temperature in the heat exchanger. The greater the flow, the lower the temperature difference between the inlet and the outlet is large at constant power. Low mean temperatures in heat exchangers lead to higher performances on the electrical efficiency and on the condensation gains.

These trends are confirmed by experimental investigations. The thermal output can be almost considered as constant, there is no modulation here. To maintain the thermal output as constant (Fig. 3), the combustion power increases with the inlet water temperature to compensate the latent heat losses which are greater (Fig. 4). The electrical power increases with low inlet water temperatures and high water volume flow rates (Fig. 5). The condensation gains increase with low inlet water temperatures and high volume flow rates (Fig. 6). The exhaust gas temperatures increase with high inlet water temperatures and slightly decrease with higher water volume flow rates (Fig. 7).

For the electrical power, the condensation gains and the exhaust gas temperature, the graphs use the volume mass flow for the x-axis to highlight the volume flow rate impact on these quantities which is less significant than the inlet water temperature.

The condensation gains are assessed at the outlet of the device by measuring manually the condensates mass during a given time that gives a mean condensate mass flow rate. The sensible heat is not taken into account, only the latent heat is used to calculate the power recovery made on the condensation of the steam in exhaust gas.

3.2.2. Additional gas boiler in steady state

The same tests have been carried out on the additional boiler. The additional gas condensation boiler can modulate its thermal power. Figs. 8–11 give the energy performances of the additional boiler. The tests use a water flow rate range different from the Stirling engine tests because the involved powers are higher. To avoid high temperature differences and too high outlet water temperatures, the nominal volume flow rate of the Stirling engine

(which also is the maximum value of the tested range for the Stirling engine) is used as the minimal volume flow rate for the additional boiler. The boiler thermal and fuel outputs are more sensitive to the inlet water temperature than the Stirling engine (Figs. 8 and 9). The volume flow rate also has a significant impact, linked to the internal control of the device. These two quantities decrease with high inlet water temperatures and low water volume flow rates. The additional boiler follows the same trends than the Stirling engine for the condensation gains and the exhaust gas temperatures (Figs. 10 and 11).

3.2.3. Transient states

The different transient phases have been studied. The results have been normalized in order to identify the general behaviour for any configurations and power levels (Figs. 12–17). The experimental data are fitted by simple laws like exponential or linear laws. Different tests have been carried out by changing the inlet water temperature and the volume flow rates. The thermal behaviour for the Stirling engine is more variable according to the test

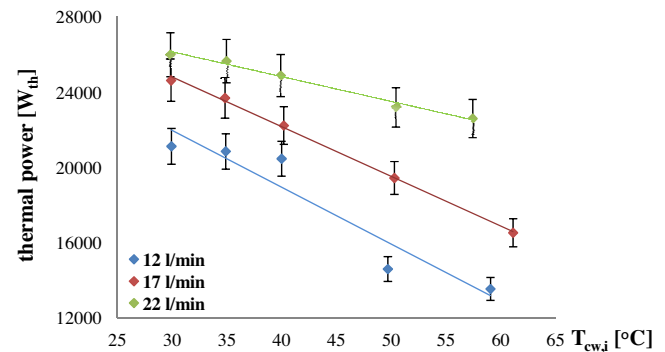


Fig. 8. Thermal power of the additional boiler.

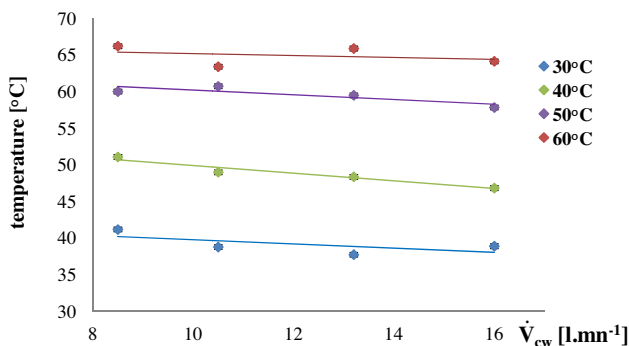


Fig. 7. Exhaust gas temperature of the Stirling engine.

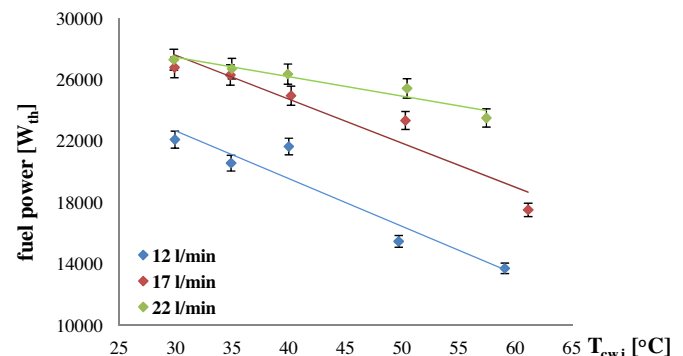


Fig. 9. Fuel power of the additional boiler.

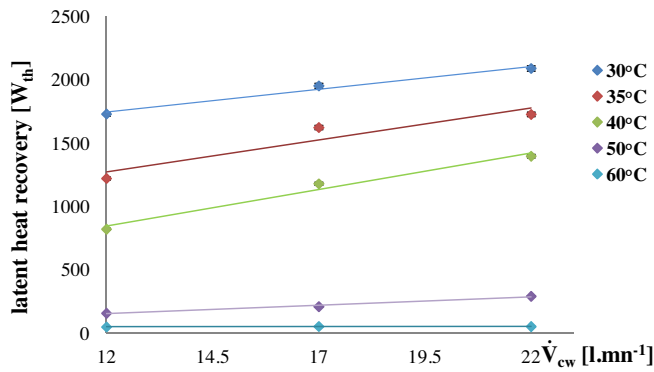


Fig. 10. Condensation gains of the additional boiler.

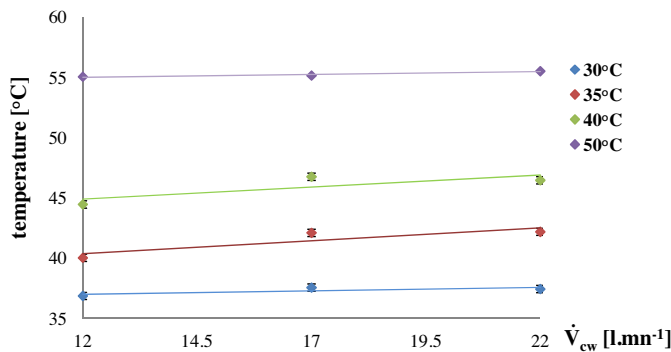


Fig. 11. Exhaust gas temperature of the additional boiler.

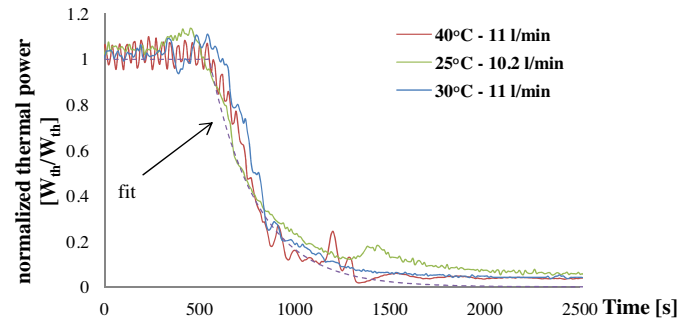


Fig. 13. Normalized thermal power during the cooling phase of the Stirling engine.

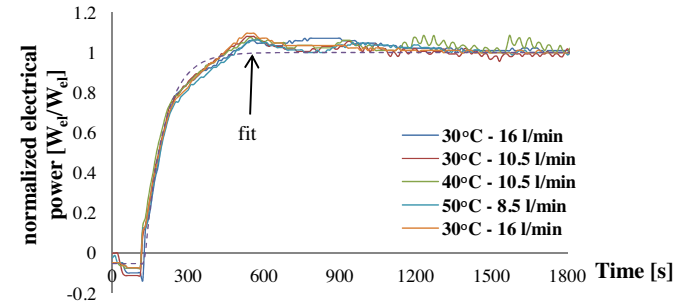


Fig. 14. Normalized electrical power during the start-up phase of the Stirling engine.

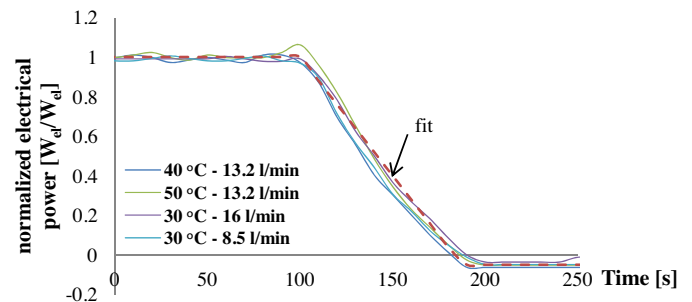


Fig. 15. Normalized electrical power during the cooling phase of the Stirling engine.

conditions. The internal control and the start-up gas over-supplies make the thermal power variable during this phase. However, to keep a consistency of the model, the general trend of the thermal output can be fitted by an exponential law in this phase by maintaining an energy equivalency (Fig. 12). The cooling phase is more stable with a similar exponential drop trend for different test conditions (Fig. 13). The electrical production has a similar behaviour during the start-up and cooling phases for any inlet water temperatures or water volume flow rates (Figs. 14 and 15). The drop is linear for the electrical production during the cooling phase (Fig. 15). The behaviour of the thermal power during the start-up and cooling phases for the additional boiler is stable for each test condition and exponential laws are used to fit the experimental

data (Figs. 16 and 17). These fittings are used later in the model development.

The tested μ CHP has only one electrical wire for outgoing and incoming electricity. It is impossible to dissociate the gross and net

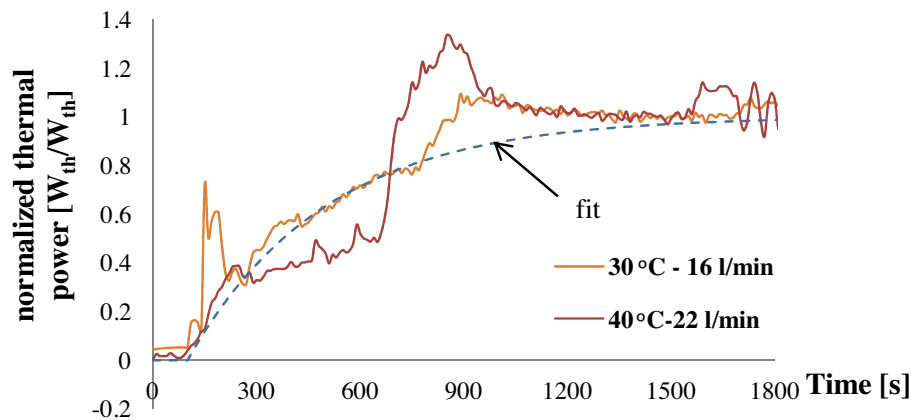


Fig. 12. Normalized thermal power during the start-up phase of the Stirling engine.

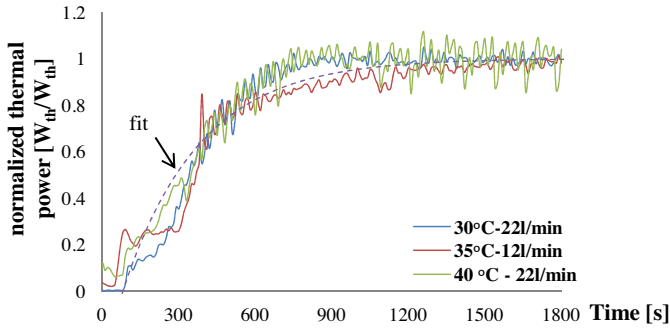


Fig. 16. Normalized thermal power during the start-up phase of the additional boiler.

production in steady state. So, the cooling phase is used to estimate the net electrical power: after the stop of the Stirling engine electrical production, only the pump, the fan and the electronic consumption are operating. Three different cooling phases are presented in Fig. 13. These tests also lead to the determination of the cooling phase duration.

Up to $\Delta t = 90$ s from the beginning of the cooling phase, the engine keeps producing electricity with a linear drop (Fig. 15). A desynchronisation appears when the head temperature decreases below 170°C (without electrical production, the net electrical power becomes negative). Up to $\Delta t = 750$ s, the fan and the pump keep working to ensure the engine cooling until a head temperature of 130°C . The measured consumed power is between 40 and 50 W (the resolution display is 10 W). A hand-held wattmeter has been used in order to get more accurate data of 44 W for the global auxiliary consumption. From $\Delta t = 750$ s, the μCHP is stopped; only the electronic power is consumed for controllers. Manual measurements gave an electronic power of 4 W.

4. Dynamic model development and identification procedure

The dynamic model and the identification procedure are based on four operation periods: start-up phase, steady state, cooling phase, and stop phase. Three kinds of parameters are distinguished: steady state parameters, transient parameters and operation logic parameters which depend on the own control strategy of the unit and which are integrated in the transient parameters (Appendix A).

The requirements of the developed model to be implemented in building energy simulation tools are summarized:

- The time step has to be equivalent to building simulation time steps: some minutes. This point excludes accurate thermodynamic models that need tiny time steps (some seconds or less).

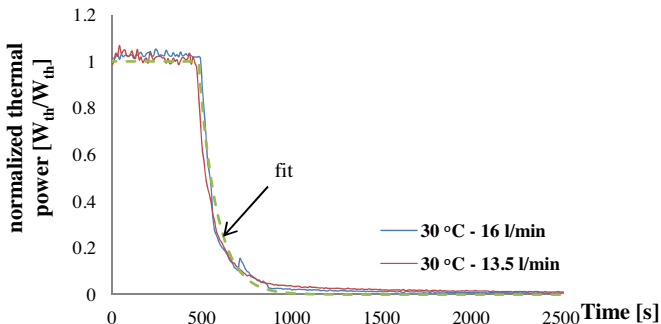


Fig. 17. Normalized thermal power during the stop phase of the additional boiler.

- The number of parameters to identify should be limited (an easy identification based on experiments is aimed where each parameter keeps a physical sense).
- The model be adaptable to other μCHP units.
- The model should be representative of the μCHP units performances (in particular, thermal and electrical transient behaviours of μCHP during the start-up and cooling phases).

In the following, the modelling is split in two parts according to the operating modes:

- Steady state.
- Transient states (start-up, cooling, stop).

4.1. Steady state

In steady state conditions, the electrical power, the fuel consumption and the heating power are assessed by correlations dependent on the inlet water temperature $T_{cw,i}$, the water flow rate $\dot{m}_{cw}$, and the thermal load in the case of modulating μCHP as proposed by Lombardi et al. [22] and Andlauer [29]. CO_2 emissions are deduced from the fuel consumption. The steady state performance is characterized by linear functions for the thermal power and the electrical power outputs and the fuel thermal input. The linear coefficients are identified from experimental tests in steady state conditions.

In the following, the equations are written for non-modulating engines (ON-OFF operation) since the tested μCHP cannot modulate his load. In case of engines able to operate at variable capacity, the correlations of the net electric power and the fuel consumption should integrate a load factor on the thermal power accounting for performance degradation/improvement with the modulation.

The expressions of the delivered thermal output, fuel input and gross electrical power, are defined in functions of the nominal conditions of flow rate and temperature. As shown in Figs. 3–5, and as proposed by Lombardi et al. [22], the equations can be written as a linear combination of water flow rate and inlet water temperature:

$$P_{\text{fuel}} = P_{\text{fuel}}^{\text{nom}} + a (T_{cw,i} - T_{cw,i}^{\text{nom}}) + b (\dot{m}_{cw} - \dot{m}_{cw}^{\text{nom}}) \quad (3)$$

$$\dot{Q}_{HX} = \dot{Q}_{HX}^{\text{nom}} + c (T_{cw,i} - T_{cw,i}^{\text{nom}}) + d (\dot{m}_{cw} - \dot{m}_{cw}^{\text{nom}}) \quad (4)$$

$$P_{\text{gross}} = P_{\text{gross}}^{\text{nom}} + e (T_{cw,i} - T_{cw,i}^{\text{nom}}) + f (\dot{m}_{cw} - \dot{m}_{cw}^{\text{nom}}) \quad (5)$$

If reference characteristics for the temperature or mass flow rate are given by the manufacturer, it is suitable to use it as nominal conditions. Otherwise, a method is given here to determine relevant nominal conditions. Concerning the nominal electrical output, the manufacturer value is used. For the inlet cooling water temperature, it is relevant to use a low temperature in order to optimize the condensation gains: 30°C for example. The nominal mass flow rate has to match the temperature difference given by the manufacturer when the μCHP operates without the additional boiler. Here a temperature difference of 5 K has been fixed that led to a nominal mass flow rate of 16 kg mn^{-1} .

In order to parameterize Eqs. (3)–(5) according to the cooling water mass flow rate and temperature, several tests should be achieved. The experiment plan consists in changing one parameter on the full test range while keeping the others in nominal conditions. The proposed ranges for the Stirling engine are compiled in Table 3.

Net electric power corresponds to the usable part of electricity produced by the engine. Another part of the production is auto consumed by the auxiliaries (pumps, fans) P_{aux} and the electronic card P_{electro} . We define:

$$P_{\text{gross}} = P_{\text{net}} + P_{\text{aux}} + P_{\text{electro}} \quad (6)$$

Table 3
Tests list for the parameterization procedure and validation for the Stirling engine.

$\dot{m}_{cw}$	$0.4 \dot{m}_{cw}^{nom}$	$0.6 \dot{m}_{cw}^{nom}$	$0.8 \dot{m}_{cw}^{nom}$	$\dot{m}_{cw}^{nom}$
$T_{cw,i}$				
$T_{cw,i}^{nom}$				
$T_{cw,i}^{nom} + 10$				
$T_{cw,i}^{nom} + 20$				
$T_{cw,i}^{nom} + 30$				

Electrical consumptions linked to the operation of the auxiliary devices like the fans and the pumps can be determined by different ways: either the μ CHP is equipped by two different electrical wires (one for the incoming electricity and one for the outgoing electricity) or these consumptions can be deduced during the cooling phase where no electricity is produced but the auxiliary devices are still working. These values can also be found in manufacturer data.

For each test, the mean value of the measurements during the steady state is taken into account. The steady state is reached when the following conditions are obtained:

$$(T_{cw,i}^{SP} - 1.5) < T_{cw,i}^{PV} < (T_{cw,i}^{SP} + 1.5) \text{ and } 0.95 \times P_{gross}^{SP} < P_{gross}^{PV} < 1.05 \times P_{gross}^{SP}$$

As shown in Figs. 9 and 10, the same equations as for the Stirling engine can be used:

$$P_{fuel\ aux} = P_{fuel\ aux}^{nom} + a' (T_{cw,i} - T_{cw,i}^{nom}) + b' (\dot{m}_{cw} - \dot{m}_{cw}^{nom}) \quad (7)$$

$$\dot{Q}_{HX\ aux} = \dot{Q}_{HX\ aux}^{nom} + c' (T_{cw,i} - T_{cw,i}^{nom}) + d' (\dot{m}_{cw} - \dot{m}_{cw}^{nom}) \quad (8)$$

The parameterization procedure is similar to those used for μ CHP. The identified parameters on the tested μ CHP boiler are given in Appendix A.

Assuming the combustion is complete, the fuel mass flow rate can be assessed from the fuel thermal input as:

$$\dot{m}_{fuel} = \frac{P_{fuel}}{GHV} \quad (9)$$

The model can be used for different types of fuel. For the model identification, the fuel mass flow rate should be measured and the gross heating value (GHV) of the fuel should be known or measured. The air excess should be also known. From the fuel mass flow rate, the fuel composition and the air excess, typical combustion calculations are undertaken in order to assess the combustion reactants mass flow rates and the exhaust gas heat losses. In addition to the steady state characterization, the start-up and stop transient phases are to be modelled.

4.2. Transient phases

The use of exponential curves for transient phases gives a good representation of the evolution during these phases even if the curves are more complex due to the desynchronisation of the power generator during the cooling phase and fuel overshooting during the preheating phase.

4.2.1. Start-up transient

The start-up phase differs according to the engine type. The following approach can be only partly considered as generic since it has been developed for the specific case of a Stirling engine. For other technologies, the model has to be adapted. Indeed, a fuel overshooting for start-up can appear. Moreover, when the engine is not fully cooled down at the ambient temperature, the start-up is partial. Andlauer [29] shows that the partial start-up mode can be considered as a full start-up mode on the specific Stirling engine studied in this paper.

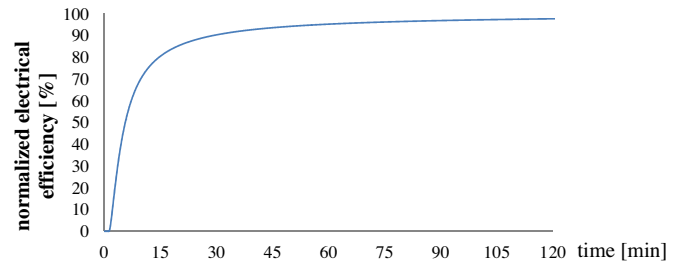


Fig. 18. Normalized electrical cycle efficiency according to the cycle duration.

During the Stirling engine start-up phase, we distinguish three phases. At the beginning, the Stirling engine head has to be heated. In a natural gas engine, this heating phase is achieved by a gas burner with a fuel mass flow rate greater than nominal mass flow in order to quickly reach the suitable temperature. In the model, no overshooting is assumed and the fuel mass flow rate is equal to the nominal conditions. As soon as the head reaches the set temperature, the free piston begins to move and the electrical production can begin. Synchronized with the network, the μ CHP produces the electricity.

We assume a first order behaviour for the electrical power, heating power and head temperature with phase shifts as experimental investigations show.

- During Start-up 1, the head temperature rises with gas combustion in the burner. We assume heating production is insignificant.
- During Start-up 2, the combustion continues and the heating production starts, except for the electrical production.
- During Start-up 3, the combustion and the heating production continue and the electrical production starts.

The heating and electrical productions during the start-up phase are assumed to increase as an exponential function using delay times and time constants (Eqs. (10) and (11)).

$$\dot{Q}_{HX} = \begin{cases} 0 & \text{for } t < t_{start} + \Delta t_{start}^Q \\ \dot{Q}_{HX}^{nom} \left(1 - e^{-\frac{t - (t_{start} + \Delta t_{start}^Q)}{\tau_{start}^Q}} \right) & \text{for } t \geq t_{start} + \Delta t_{start}^Q \end{cases} \quad (10)$$

$$P_{net} = \begin{cases} 0 & \text{for } t < t_{start} + \Delta t_{start}^Q \\ P_{net}^{nom} \left(1 - e^{-\frac{t - (t_{start} + \Delta t_{start}^P)}{\tau_{start}^P}} \right) & \text{for } t \geq t_{start} + \Delta t_{start}^P \end{cases} \quad (11)$$

These parameters can be identified by experimental investigations (Section 3.2.3). Physical models exist by taking into account the inertia of the heat exchanger or the mass of water [30], but the goal here is to propose a simple approach of modelling which requires the less parameters as possible. The identified parameters are given in the Appendix A. The description of the modelling of the start-up phase is given in Appendix B.

The modelling of the transient phase lets to take into account the temporal inertia of these systems which have to match the heat and power demands. The profitability of these systems is linked to the electrical efficiency but also to the duration of the cycle. Fig. 18 gives the evolution of the ratio between the electrical cycle efficiency according to the cycle duration and the nominal efficiency in steady state (normalized electrical efficiency). It shows that to reach high-normalized electrical efficiencies (95% of the nominal value for example), the system has to operate at least 1 h.

4.2.2. Cooling transient

When the combustion stops inside the μ CHP engine, the pumps and fans keep operating for a given period called “cooling”, in order to use the residual heat and to cool down the engine. Then, these auxiliary devices are switched off. The heating production drop is approximated by a first order transfer function, characterized by a time constant (Eq. (12)). A partial cooling mode could be taken into account but it is not implemented in the model developed.

$$\dot{Q}_{HX} = \begin{cases} 0 & \text{for } t > t_{stop} + \Delta t_{stop}^Q \\ \dot{Q}_{HX}^{nom} e^{-\frac{t-t_{stop}}{\tau_{stop}^Q}} & \text{for } t \leq t_{stop} + \Delta t_{stop}^Q \end{cases} \quad (12)$$

$$P_{net} = \begin{cases} -P_{aux} - P_{electro} & \text{for } t > t_{stop} + \Delta t_{stop}^Q \\ P_{gross}^{nom} \left(1 - \frac{t-t_{stop}}{\Delta t_{stop}^Q}\right) - P_{aux} - P_{electro} & \text{for } t \leq t_{stop} + \Delta t_{stop}^Q \end{cases} \quad (13)$$

The time constants are to be identified on cooling phases (Figs. 13 and 15).

The μ CHP remains able to produce electricity thanks to the accumulated heat on the hot side of the engine. The electrical production lasts a short period compared to the pump and fan operation time for the engine cooling. The electrical output decreases until 0, but not necessary exponentially (Fig. 13). The proposed method consists in fitting the electric drop by a linear law (Eq. (13)).

The cooling phase period, while the fan and the pump keep operating, depends on engine control: either the engine head temperature reaches a limit or a certain time is elapsed. The model uses a time limit, after which the cooling phase stops. This choice is adapted for the tested Stirling engine since the engine is non-modulating, so that the head temperature is always the same at the beginning of the cooling phase and thus the cooling time.

During the engine stop phase, after the cooling phase, the fan and the pump are stopped. The engine temperature evolution is only governed by heat exchanges with the ambience $\dot{Q}_{loss}$, the thermal, fuel and electrical powers are equal to 0. Only the electronic power $P_{electro}$ is consumed.

The model parameters for the cooling phase of the tested μ CHP are presented in Appendix A. The modelling approach of the cooling phase is described in the Appendix B.

4.3. Exhaust gas heat loss

4.3.1. Exhaust gas temperature

The exhaust gas temperature is modelled with linear correlations using the cooling water mass flow rate and temperature (Figs. 7 and 11), like the other main inputs or outputs.

$$T_{exh} = g (T_{cw,i} - T_{cw,i}^{nom}) + h (\dot{m}_{cw} - \dot{m}_{cw}^{nom}) + T_{exh}^{nom} \quad (14)$$

$$T_{exh,aux} = g' (T_{cw,i} - T_{cw,i}^{nom}) + h' (\dot{m}_{cw} - \dot{m}_{cw}^{nom}) + T_{exh,aux}^{nom} \quad (15)$$

The model parameters for the exhaust gas temperature of the tested μ CHP are presented in Appendix A.

4.3.2. Condensation gains

The condensation gains are modelled by linear correlations. From Figs. 6 and 10, it can be defined as:

$$\dot{m}_{cond} = \max (0; \dot{m}_{cond}^{nom} + i (T_{cw,i} - T_{cw,i}^{nom}) + j (\dot{m}_{cw} - \dot{m}_{cw}^{nom})) \quad (16)$$

$$\dot{m}_{cond,aux} = \max (0; \dot{m}_{cond,aux}^{nom} + i' (T_{cw,i} - T_{cw,i}^{nom}) + j' (\dot{m}_{cw} - \dot{m}_{cw}^{nom})) \quad (17)$$

The model parameters for the condensation gains of the tested μ CHP are presented in Appendix A.

4.3.3. Enthalpy flows

The exhaust gas latent heat losses, that is to say the latent energy of the non-condensed fraction of steam are defined in Eq. (18).

$$\dot{H}_{exh}^{lat} = (\dot{m}_{cond}^{max} - \dot{m}_{cond}) L_v \quad (18)$$

The maximum condensation mass flow rate is assessed as the mass flow rate of the steam in the exhaust gas based on the fuel composition and the fuel mass flow rate. The reactants and combustion products enthalpies are calculated from JANAF tables [31] giving for each component a polynomial expression of the specific enthalpy.

$$\dot{H}_i = \frac{\dot{m}_i}{M_i} \sum_i \{ [H_i(T) - H_i(T_0)] \chi_i \} \quad (19)$$

Finally, the exhaust flow rate becomes:

$$\dot{m}_{exh} = \dot{m}_{fuel} + \dot{m}_{air} - \dot{m}_{cond} \quad (20)$$

With:

$$\dot{m}_{air} = \dot{m}_{fuel} V_{air} (1 + e_{air}) \frac{M_{air}}{V_m} \quad (21)$$

4.4. Dynamic model

If the main outputs of the model are given by the correlations presented above, the use of a dynamic model provides information on the behaviour of the μ CHP such as the repartition of heat losses and dynamic phases of the engine. The proposed model uses only one thermal capacitance for the whole μ CHP device. This assumption is made to facilitate the model parameterization; the start-up and cooling periods are each represented by only one time constant. A fictitious “global average internal temperature” T_{int} is introduced which governs the heat losses between the μ CHP skin and the ambience using a fictitious coefficient $[UA]_{loss}$.

The energy balance can be written as:

$$[MC]_{int} \cdot \frac{dT_{int}}{dt} = \dot{H}_{fuel} + \dot{H}_{air} + P_{fuel} - P_{gross} - \dot{Q}_{HX} - \dot{Q}_{loss} - \dot{H}_{exh}^{sen} - \dot{H}_{exh}^{lat} \quad (22)$$

$$\dot{Q}_{loss} = [UA]_{loss} \cdot (T_{int} - T_{amb}) \quad (23)$$

Due to the conventional definition of T_{int} , $[UA]_{loss}$ is strictly relative to its assessment. A simple formula giving T_{int} based on thermal measurements (based on the inlet cooling water temperature and the hottest temperature measured on the hot side of the Stirling engine) is proposed by Andlauer and al [29,32]. The first order differential equation can be solved at each time step by using a finite difference method.

The model is presented in a flow diagram in the Appendix C.

5. Simplified model

A model which can extrapolate the system performances in different operation ranges even when the control strategy or the design changes is very useful and relevant provided that the number of parameters is limited and their identification requires few experimental investigations [33]. The model of the Stirling engine needs to identify 28 parameters which can be estimated by experimental investigations as shown in Section 4. In this part, a reduction of the model parameter number is proposed in order to facilitate the model parameterization.

Table 4
Comparison between the model and experiments during steady state for the Stirling engine.

Complete method (%)	$T_{cw,i}, ^\circ\text{C}$	29.8	40.2	50.1	60.0	29.9	40.4	50.2	60.0	29.8	40.5	50.3	59.9
	$\dot{m}_{cw} \text{ l s}^{-1}$	8.5	8.5	8.5	8.5	13.2	13.2	13.2	13.2	16	16	16	16
	P_{fuel}	-1.3	-0.3	-2.5	3.2	-0.1	1.1	-3.3	0.3	3.4	-0.5	-2.4	0.4
	$\dot{Q}_{HX}$	-3.1	0.1	2.8	-0.6	-1.2	1.2	0.9	-1.6	2.7	-0.6	-0.9	-1.1
	P_{gross}	0.6	-0.9	-1.1	1.4	1.1	-0.8	-1.2	0.8	0.7	-0.1	-1.2	0.6
Simplified method (%)	T_{exh}	-0.6	-1.3	-0.9	3.4	4.9	1.6	-2.4	1.7	1.3	3.4	-0.9	1.2
	P_{fuel}	-2.7	-1.7	-3.9	1.8	0.4	1.6	-2.9	0.8	5.1	1.1	-0.9	2.0
	$\dot{Q}_{HX}$	-2.9	0.5	3.5	0.5	-1.8	0.9	0.9	-1.4	1.7	-1.4	-1.4	-1.3
	P_{gross}	0.9	-0.6	-0.8	1.7	1.0	-0.9	-1.3	0.7	0.4	-0.5	-1.5	0.2
	T_{exh}	-8.3	-5.6	-3.2	2.7	0.4	0.0	-2.2	3.3	-0.9	3.6	0.9	4.0

Table 5
Comparison between model and experiments during steady state for the additional boiler.

Complete method (%)	$T_{cw,i}, ^\circ\text{C}$	29.9	40	49.7	59.1	29.9	40.2	50.3	61.1	29.8	39.9	50.4	57.4
	$\dot{m}_{cw} \text{ l s}^{-1}$	12	12	12	12	17	17	17	17	22	22	22	22
	$P_{fuel \text{ aux}}$	5.9	-3.8	18.6	16.3	-1.5	-4.9	-9.3	4.9	7.6	1.6	-5.3	-5.1
	$\dot{Q}_{HX \text{ aux}}$	0.3	-8.8	11.6	5.4	-1.1	-1.4	0.5	2.6	5.5	1.0	-2.4	-7.7
	$T_{exh \text{ aux}}$	2.5	0.4	-6.9	2.8	1.7	-3.4	-5.6	3.4	2.9	-2.3	-5.4	5.9

Table 6
Comparison between model and experiments during transient states for the Stirling engine.

Start-up (%)	$T_{cw,i}, ^\circ\text{C}$	29.8	40.2	50.1	60.0	29.9	40.4	50.2	60.0	29.8	40.5	50.3	59.9
	$\dot{m}_{cw} \text{ l s}^{-1}$	8.5	8.5	8.5	8.5	13.5	13.5	13.5	13.5	16	16	16	16
	$\dot{Q}_{HX}$	12.0	-10.3	-9.2	13.3	-12.3	-12.4	8.7	9.2	-9.2	10.0	-7.3	-6.3
Cooling (%)	P_{gross}	0.9	-0.6	1.1	0.8	-1.8	1.1	-2.0	1.3	-1.5	0.4	-1.0	-0.3
	$\dot{Q}_{HX}$	1.2	2.1	3.2	-1.2	-1.8	4.1	-1.3	2.0	3.4	1.1	-2.7	0.9
	P_{gross}	5.3	4.2	-2.3	3.7	-1.0	2.0	-4.2	1.7	-6.1	-2.1	5.0	2.4

Table 7
Comparison between model and experiments during transient states for the additional boiler.

Start-up (%)	$T_{cw,i}, ^\circ\text{C}$	29.8	40.2	50.1	60.0	29.9	40.4	50.2	60.0	29.8	40.5	50.3	59.9
	$\dot{m}_{cw} \text{ l s}^{-1}$	12	12	12	12	17	17	17	17	22	22	22	22
Cooling (%)	$\dot{Q}_{HX \text{ aux}}$	-8.2	1.2	2.7	-5.0	5.1	-7.6	3.0	-5.8	2.8	-9.7	-4.1	5.6
	$\dot{Q}_{HX \text{ aux}}$	6.6	9.2	-2.1	2.2	-11.9	4.2	-7.0	4.6	-6.0	7.2	-1.1	7.0

5.1. Reduction of the number of parameters

Eqs. (3)–(5), (14), and (16) for the fuel power, the heating power, the electrical power, the exhaust gas temperature and the condensation gains can be simplified. As shown in Figs. 3, 4, 19 and 20, the influence of the cooling water mass flow rate can be neglected and the influence of the inlet cooling water temperature, which is more significant, can be characterized by a linear function. For the heating power, it can be simplified as a constant value (Fig. 3).

The different equations become:

$$P_{fuel} = P_{fuel}^{nom} + a (T_{cw,i} - T_{cw,i}^{nom}) \quad (24)$$

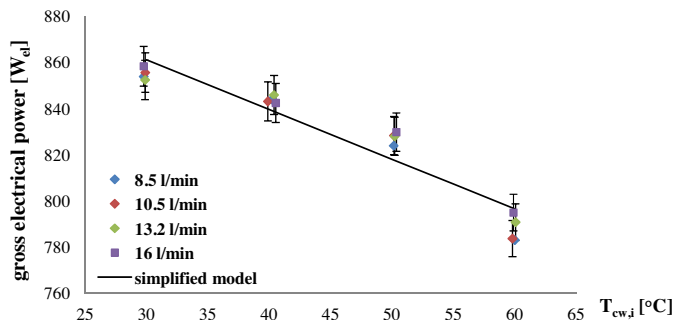


Fig. 19. Gross electrical power of the Stirling engine.

$$\dot{Q}_{HX} = \dot{Q}_{HX}^{nom} \quad (25)$$

$$P_{gross} = P_{gross}^{nom} + e (T_{cw,i} - T_{cw,i}^{nom}) \quad (26)$$

$$T_{exh} = T_{cw,i} + \Delta T \quad (27)$$

The last simplification consists to include all the heat losses (latent and sensible exhaust gas heat losses, skin heat losses, air and fuel enthalpy flows) in one heat loss source limiting the information provided by the model but avoiding an experimental characterization of the condensate mass flow rate.

$$\dot{H}_{loss} = P_{fuel} - P_{gross} - \dot{Q}_{HX} \quad (28)$$

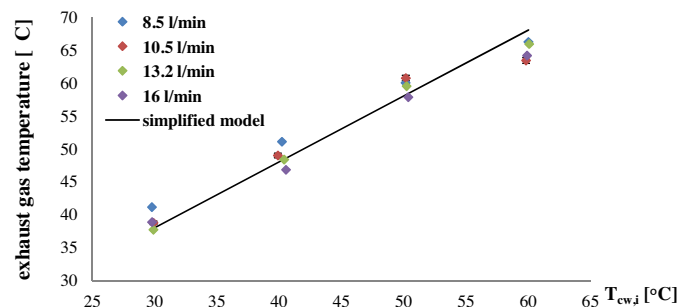


Fig. 20. Exhaust gas temperature of the Stirling engine.

5.2. Use of manufacturer data

With these simplifications, the model requires to identify only 17 parameters. Some manufacturer data are generally available and could be used to identify several parameters. Main behaviour laws are linear so the modelling just needs two performance points to characterize the unit. It seems reasonable to get performance data from manufacturers for two operation points at two different temperatures, T_1 and T_2 . The auxiliary and electronic powers of the μ CHP and the air excess can be obtained from the manufacturers. The remaining parameters such as time constants seem more difficult to obtain from manufacturers. Default values could be chosen or identified by an experimental test with a start-up and a cooling phase. The simplified model parameters are listed in [Appendix A](#). The reduction of parameters is only applied on the Stirling engine. On the additional boiler, only the simplification on the exhaust gas and the aggregation of the heat losses are used, the mass flow rate has a strong impact on the outputs and cannot be neglected.

6. Validation of the model and discussions

The model with the two-parameterization methods (28 and 17 parameters) is compared to experimental results in steady and transient states. The experimental measurements are carried out after more than 3 h of steady state operation times. The tests are carried out for different inlet cooling water temperatures and cooling water mass flow rates. [Table 4](#) gives the relative difference between the two models of μ CHP and the experimental results.

The relative differences between the simplified and the complete parameter identification procedures show that the both methods are very suitable to correctly model the real behaviour of the unit.

[Table 5](#) shows the results of the comparison between the additional gas boiler model and the experimental results.

[Tables 6 and 7](#) show the relative differences between the Stirling engine and the additional gas boiler models and the experimental results for the transient states. The modelling method of the transient phases by time constants is very suitable to correctly model the real transient behaviour of the unit except for the thermal output during the start-up of the Stirling engine where the difference is higher at about 10% because of the more chaotic behaviour.

The Stirling engine can work simultaneously with the additional boiler. In this configuration, the inlet water temperature of the additional boiler matches to the outlet water temperature of the Stirling engine. The steady state and transient inputs and outputs are added by using the superposition principle.

The efficiencies of this system decrease with higher temperatures and short operation durations. The coupling of these systems with building needs to be optimized in order to improve their energy, economic and environmental relevance.

The control strategy is made outside the model by giving the mass flow rate, the inlet water temperature and the heat loads as inputs for the model ([Appendix C](#)). The mass flow can be fixed at a constant value or be variable to match a thermal output of the auxiliary, the water inlet temperature can be given by a water logic based on outdoor temperature and the heat loads can be given by a data file or a dynamic thermal simulation of a building. Its also possible to implement “power led” or “cost led” control strategies by triggering the μ CHP unit only if there is a sufficient power demand or if it is economically profitable: the heat can be stored in a buffer tank if there is no needs, otherwise, the heat demand can be covered by the auxiliary.

7. Conclusion

A semi-physical model of a μ CHP device and a gas boiler has been developed and implemented in the TRNSYS environment in order to assess the energy, environmental and economic impacts of μ CHP. The main advantages of the model are its simplicity while keeping a good accuracy. The model is able to simulate transient phases which can be important to assess the global efficiency of the device.

The parameters have been identified by experimental investigations on a gas Stirling engine μ CHP unit. The initial method has been simplified in order to ease the identification of the model only with manufacturer data and a few experimental tests. The simplified method does not degrade the accuracy of the model. The model has been validated for the Stirling engine for inlet cooling water temperatures from 30 to 60 °C and cooling water flow rates from 8 to 16 l min⁻¹.

At the moment, this model is only adapted for gas applications: natural gas, propane or biogas. Its adaptation for biomass-CHP requires some improvements such as a CO-emission model during start-up/cooling phases or a new combustion model for the biomass fuel as Bouvenot et al. [34] did for a wood pellet μ CHP device. Moreover, the model has been only validated for a non-modulating Stirling μ CHP. Future works will be undertaken to extend the applications of this μ CHP model.

Then, these models will be used in order to optimize their use in buildings by coupling them with buildings in numerical platforms like TRNSYS. The aim will be to identify the most suitable environment and operation of these systems to maximize their efficiencies and their profitability.

Acknowledgement

The authors would like to thank De Dietrich, Enerest and Region Alsace for their support in the achievement of the project.

Appendix A. Model parameters

[Tables A1–A3.](#)

Appendix B. Transient phases modelling approach and steady state to transient states transition conditions

[Figs. B1–B3.](#)

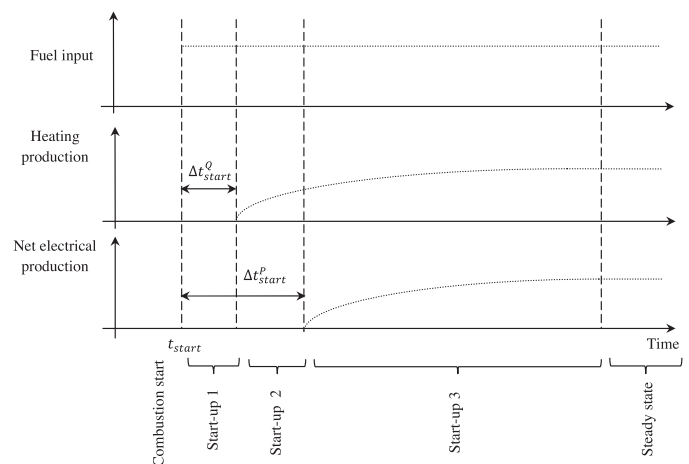


Fig. B1. Modelling approach of the start-up phase.

Table A1
Model parameters of the Stirling engine.

Steady state parameters				Transient state parameters			
	Parameter	Value	Unit		Parameter	Value	Unit
1	$\dot{m}_{cv}^{nom}$	16	kg mn^{-1}	21	τ_{start}^Q	400	s
2	$T_{cw,i}^{nom}$	30	$^{\circ}\text{C}$	22	τ_{start}^P	75	s
3	p_{fuel}^{nom}	6640	W_{th}	23	Δt_{start}^Q	45	s
4	Q_{HX}^{nom}	5315	W_{th}	24	Δt_{start}^P	75	s
5	p_{gross}^{nom}	864	W_{el}	25	τ_{stop}^Q	250	s
6	$P_{electro}$	4	W_{el}	26	Δt_{stop}^Q	750	s
7	P_{aux}	40	W_{el}	27	Δt_{stop}^P	85	s
8	a	12.35	$\text{W}_{th} \text{K}^{-1}$	28	$[MC]_{int}$	12,350	J K^{-1}
9	b	-27.44	$\text{W}_{th} (\text{kg mn}^{-1})^{-1}$				
10	c	-1.43	$\text{W}_{th} \text{K}^{-1}$				
11	d	8.62	$\text{W}_{th} (\text{kg mn}^{-1})^{-1}$				
12	e	-2.16	$\text{W}_{el} \text{K}^{-1}$				
13	f	0.73	$\text{W}_{el} (\text{kg mn}^{-1})^{-1}$				
14	T_{exh}^{nom}	38.8	$^{\circ}\text{C}$				
15	g	0.91	-				
16	h	-0.31	$^{\circ}\text{C} (\text{kg mn}^{-1})^{-1}$				
17	$\dot{m}_{cond}^{nom}$	2.16 E^{-4}	kg s^{-1}				
18	i	2.84 E^{-6}	$\text{J kg}^{-1} \text{ }^{\circ}\text{C}^{-1}$				
19	j	-8.26 E^{-6}	$\text{J kg}^{-1} (\text{kg mn}^{-1})^{-1}$				
20	$[UA]_{loss}$	2.76	W K^{-1}				

Table A2
Model parameters of the additional boiler.

Steady state parameters				Transient state parameters			
	Parameter	Value	Unit		Parameter	Value	Unit
1	$\dot{m}_{cv}^{nom}$	16	kg mn^{-1}	18	$\tau_{start aux}^Q$	450	s
2	$T_{cv,i}^{nom}$	30	$^{\circ}\text{C}$	19	$\Delta t_{start aux}^Q$	45	s
3	$p_{fuel aux}^{nom}$	25,769	W_{th}	20	$\tau_{stop aux}^Q$	100	s
4	$Q_{HX aux}^{nom}$	23,703	W_{th}	21	$\Delta t_{stop aux}^Q$	750	s
5	$P_{electro}$	4	W_{el}	22	$[MC]_{int}$	12,350	J K^{-1}
6	P_{aux}	60	W_{el}				
7	a'	-256.1	$\text{W}_{th} \text{K}^{-1}$				
8	b'	594.14	$\text{W}_{th} (\text{kg mn}^{-1})^{-1}$				
9	c'	-236.2	$\text{W}_{th} \text{K}^{-1}$				
10	d'	629.88	$\text{W}_{th} (\text{kg mn}^{-1})^{-1}$				
11	$T_{exh aux}^{nom}$	38.2	$^{\circ}\text{C}$				
12	g'	0.68	-				
13	h'	0.08	$^{\circ}\text{C} (\text{kg mn}^{-1})^{-1}$				
14	$\dot{m}_{cond aux}^{nom}$	7.58 E^{-4}	kg s^{-1}				
15	i'	1.63 E^{-5}	$\text{J kg}^{-1} ^{\circ}\text{C}^{-1}$				
16	j'	-3.37 E^{-5}	$\text{J kg}^{-1} (\text{kg mn}^{-1})^{-1}$				
17	$[UA]_{loss}$	2.76	W K^{-1}				

Table A3
Simplified model parameters of the Stirling engine.

Steady state parameters				Transient state parameters			
	Parameter	Value	Unit		Parameter	Value	Unit
1	$\dot{m}_{cw}^{nom}$	16	kg mn^{-1}	11	τ_{start}^Q	400	s
2	$T_{cw,i}^{nom}$	30	$^{\circ}\text{C}$	12	τ_{start}^P	75	s
3	p_{fuel}^{nom}	6,749	W_{th}	13	Δt_{start}^Q	45	s
4	$\dot{Q}_{HX}^{nom}$	5,260	W_{th}	14	Δt_{start}^P	75	s
5	p_{gross}^{nom}	861	W_{el}	15	τ_{stop}^Q	250	s
6	$P^{electro}$	4	W_{el}	16	Δt_{stop}^Q	750	s
7	P_{aux}	40	W_{el}	17	Δt_{stop}^P	85	s
8	a	12.32	$\text{W}_{th}\text{K}^{-1}$				
9	e	-2.16	$\text{W}_{el}\text{K}^{-1}$				
10	ΔT	8	$^{\circ}\text{C}$				

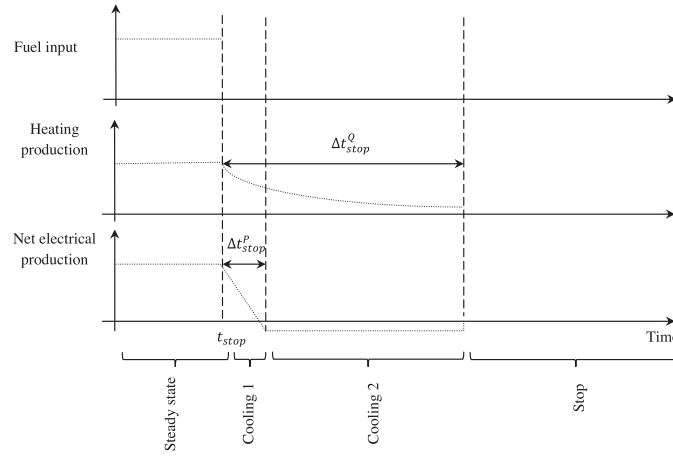
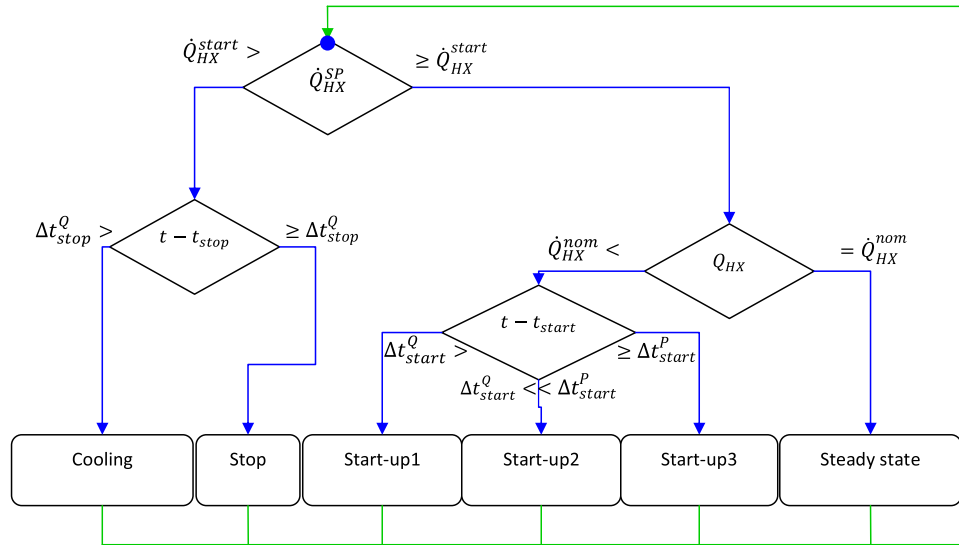


Fig. B2. Modelling approach of the cooling phase.



Current mode	Future mode	Trigger
Stop	Start-up1	$\dot{Q}_{HX}^{SP} \geq \dot{Q}_{HX}^{start}$
Start-up1	Start-up2	End for $t - t_{start} > \Delta t_{start}^Q$
Start-up2	Start-up3	End for $t - t_{start} > \Delta t_{start}^P$
Start-up3	Steady state	$\dot{Q}_{HX}(t) = \dot{Q}_{HX}^{nom}$
Steady state	Cooling	$\dot{Q}_{HX}^{SP} < \dot{Q}_{HX}^{start}$
Cooling	Stop	End for $t - t_{stop} > \Delta t_{stop}^Q$

Fig. B3. steady state to transient states transition conditions.

Appendix C. Algorithm chart

Fig. C1.

- internal parameters surrounded in dotted orange line
- 28 external parameters to be identified surrounded in solid orange line and written in black

- 17 external parameters to be identified of the simplified model surrounded in solid orange line and written in violet
- parameters which can be internal (default values) or external (to be identified) surrounded in solid red line
- main outputs surrounded in blue line
- main inputs surrounded in black line

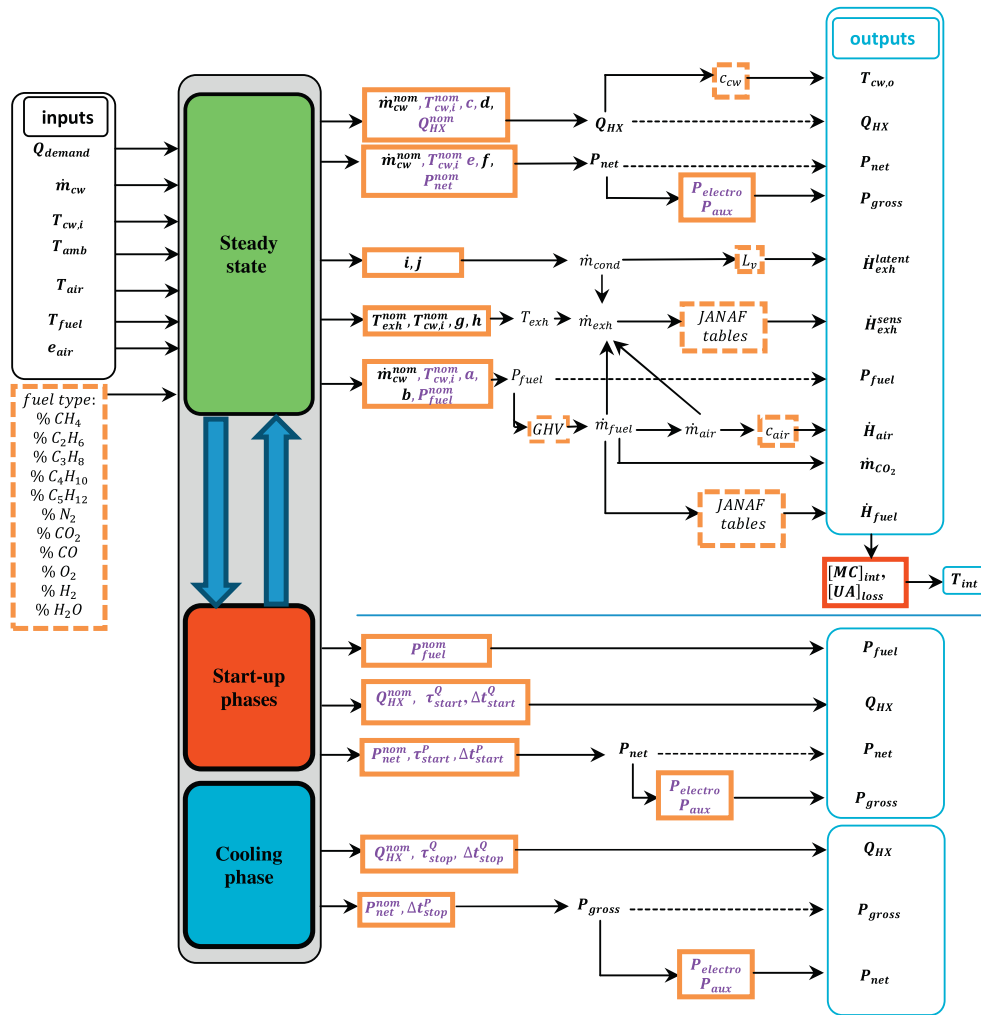


Fig. C1. Algorithm flow chart.

References

- [1] M. Peht, Environmental impacts of distributed energy systems – the case of micro cogeneration, *Environmental Science and Policy* 11 (February (1)) (2008) 25–37.
- [2] A. Dorer, A. Weber, Energy and carbon emission footprint of micro-CHP systems in residential buildings of different energy demand levels, *Journal of Building Performance Simulation* 2 (1) (2009) 31–46.
- [3] L. Spitalny, J.M.A. Myrzi, T. Melhlhorn, Estimation of the economic addressable market of micro CHP and heat pumps based on the status of the residential building sector in Germany, *Applied Thermal Engineering* (2014).
- [4] S.P. Borg, N.J. Kelly, High resolution performance analysis of micro-trigeneration in an energy-efficient residential building, *Energy and Buildings* 67 (12) (2013) 153–165.
- [5] C. Vuillecard, C.E. Hubert, R. Contreau, A. Mazzenga, P. Stabat, J. Adnot, Small-scale impact of gas technologies on electric load management – μ CHP & hybrid heat pump, *Energy* 36 (2011) 2912–2923.
- [6] M. Bianchi, A. De Pascale, P.R. Spina, Guidelines for residential micro-CHP systems design, *Applied Energy* 97 (2012) 673–685.
- [7] H.I. Onovwiona, V.I. Ugursal, Residential cogeneration systems: review of the current technology, *Renewable and Sustainable Energy Reviews* 10 (October (5)) (2006) 389–431.
- [8] S. Schulz, F. Schwendig, A general simulation model for stirling cycles, *Journal of Engineering for Gas Turbines and Power* 118 (January (1)) (1996) 1–7.
- [9] B. Kongtragool, S. Wongwises, A review of solar-powered Stirling engines and low temperature differential Stirling engines, *Renewable and Sustainable Reviews* 7 (2003) 131–154.
- [10] A.J. Organ, *The Regenerator and the Stirling Engine*, Mechanical Engineering Publications Limited, London, 1997.
- [11] K. Mahkamov, Design improvements to a biomass Stirling engine using mathematical analysis and 3D CFD modeling, *Journal of Energy Resources Technology* (2006) 128–203.
- [12] I. Urieli, D.M. Berchowitz, *Stirling Cycle Engine Analysis*, Adam Hilger Ltd, Bristol, 1984.
- [13] I. Beausoleil-Morrison, N. Kelly, Specifications for Modelling Fuel Cell And Combustion-Based Residential Cogeneration Device Within Whole-Building Simulation Programs, IEA/ECBCS Annex 42 Subtask B Report, Natural Resources Canada, Ottawa, 2007.
- [14] K.A.B. McRorie, C.P. Underwood, Small scale combined heat and power simulations: development of a dynamic spark ignition engine model, *Building Services Research and Technology* (1996).
- [15] N. Kelly, Towards a Design Environment for Building-Integrated Energy Systems: The Integration of Electrical Power Flow Modelling With Building Simulation (Ph.D. thesis), University of Strathclyde, 1998.
- [16] J. Pearce, B. Al Zahawi, R. Shuttleworth, Energy generation in the home: modelling of single-house domestic combined heat and power, *IEE Proceedings of Science, Measurement and Technology* 148 (5) (2001) 197–203.
- [17] M. Bianchi, A. De Pascale, F. Melino, Performance analysis of an integrated CHP system with thermal and electric energy storage for residential application, *Applied Energy* 112 (2013) 928–938.
- [18] S. Thiers, B. Aoun, B. Peuportier, Experimental characterization, modelling and simulation of a wood pellet micro-combined heat and power unit used as a heat source for a residential building, *Energy and Buildings* 42 (6) (2010) 896–903.
- [19] Journal Officiel de la République Française, Arrêté du 11 mars 2009 relatif à l'agrément de la demande de titre V relative à la prise en compte des chaudières à micro-cogénération à combustible liquide ou gazeux dans le cadre de la réglementation thermique 2005, Paris, France, 2009.
- [20] G. Conroy, A. Duffy, L.M. Ayompe, Validated dynamic energy model for a Stirling engine μ -CHP unit using field trial data from a domestic dwelling, *Energy and Buildings* 62 (2013) 18–26.
- [21] C. Roselli, M. Sasso, S. Sibilo, P. Tzscheuschler, Experimental analysis of micro-cogenerators based on different prime movers, *Energy and Buildings* 43 (2011) 796–804.
- [22] K. Lombardi, V.I. Ugursal, I. Beausoleil-Morrison, Proposed improvements to a model for characterizing the electrical and thermal energy performance of

- Stirling engine micro-cogeneration devices based upon experimental observations, *Applied Energy* 87 (10) (October 2010) 3271–3282.
- [23] S.P. Borg, N.J. Kelly, The development and calibration of a generic dynamic absorption chiller model, *Energy and Buildings* 55 (2012) 533–544.
- [24] Carbon Trust, The Carbon Trust's Small-Scale CHP Field Trial Update, Carbon Trust, London, 2005.
- [25] AFNOR, NF EN 13203-2. Appareils domestiques produisant de l'eau chaude sanitaire utilisant les combustibles gazeux: Appareils de débit calorifique inférieur ou égal à 70 kW et de capacité de stockage inférieure ou égale à 300 litres. Partie 2: Évaluation de la consommation énergétique, AFNOR, 2006.
- [26] Greenhouse Gas Technology Center. Generic Verification Protocol – Distributed Generation and Combined Heat and Power Field Testing Protocol, Greenhouse Gas Technology Center, Southern Research Institute, Research Triangle Park, NC, 2005.
- [27] AFNOR, Norme NF ENV 13005 (X07-020), Guide pour l'expression de l'incertitude de mesure, 1999.
- [28] M. Priel, Incertitudes de mesure et tolérances, *Techniques de l'ingénieur R 285* (1999).
- [29] B. Andlauer, Optimisation systémique de micro-cogénérateurs intégrés aux bâtiments (Ph.D. thesis), Mines ParisTech, Paris, France, 2012.
- [30] M.Y. Haller, et al., A unified model for the simulation of oil, gas and biomass space heating boilers for energy estimating purposes, *Journal of Building Performance Simulation* 4 (1) (2011) 19–36.
- [31] M.-W. Chase, NIST Janaf Thermochemical Tables, American Institute of Physics, 1998.
- [32] J.-B. Bouvenot, B. Andlauer, B. Flament, P. Stabat, D. Marchio, B. Latour, M. Siroux, Modélisation numérique de solutions de micro cogénération, congrès français de thermique, Gérardmer, France, 2013.
- [33] V. Lemort, S. Bertagnolio, A Generalized Simulation Model of Chiller and Heat Pumps to be Calibrated on Published Manufacturer's Data, International Symposium in Refrigeration Technology, Zhuhai, China, 2010.
- [34] J.-B. Bouvenot, B. Latour, P. Stabat, B. Flament, M. Siroux, D. Marchio, Y. Mermond, Modélisation numérique d'une solution de micro cogénération biomasse, Congrès français de thermique, Lyon, 2014.